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Description

TECHNICAL FIELD

5 This invention relates to overrunning spring clutch assemblies which employ a helical band in the form of a coil spring as a friction element for transmitting torque from a drive member to a driven member.

BACKGROUND

10 Various spring clutches are known for transferring torque from a drive member to a driven member such as that disclosed in U.S. Patent No. 4,570,318 to Kish. Spring clutch assemblies are preferred as torque transmitting devices where overrunning conditions would be encountered. Such conditions occur where the driven member attains a higher rotational speed than the drive member and therefore declutching is required to prevent rotation of the driver. Overrunning spring clutch assemblies generally utilize a coil
15 spring which expands radially under a driven load to couple two clutch members, an input member and an output member. Each clutch member includes a bore where the coil spring is located, with approximately one half of the spring in the input member and one half in the output member. The clutch spring is usually fixed at the drive end through press fitting onto an arbor which is attached to the input member. The arbor is coaxial with the input and output members, passing through the hollow core of the spring. The arbor
20 serves as a centering means for alignment and support of the coil spring.

In the non-rotating state, a space is provided between the spring and the inner bore surfaces of the clutch members, except at the ends, where "teaser" coils are in contact with the inner surfaces. These teaser coils ride on the surfaces to actuate the spring. When the input member begins to rotate, in a direction counter to the pitch of the spring coils, the teaser coils dragging against the output member cause
25 the spring to unwind, expanding radially so that it is in driving contact with the bore surfaces along its entire length, transferring torque from the input to output member.

In the overrunning condition, the driven member rotates faster than the drive member, in the direction of the spring pitch, thereby driving the teaser coils to rewind the spring and disengage the driven member from the driver. This may occur, for example, in a helicopter after engine shutdown when momentum
30 maintains main rotor rotation.

The French patent FR-A-718 783 discloses an overrunning spring clutch assembly comprising a coil spring for torque transmission between an input drive member and an output driven member of the clutch. The input member and the output member both are placed in coaxial alignment to an arbor extending interiorly through both members. The arbor is rotatable supported by bearing means disposed in the
35 output member. The coil spring is fixed to the input member by engagement of a projection at one of its ends with a hollow in the input member.

While spring clutches have been utilized in many applications, various problems have limited their usefulness in helicopter aircraft. Attempts have been made to expand the speed range of spring clutches, typically used from 4000 to 8000 RPM, to high RPM applications of up to approximately 30,000 RPM, to
40 increase their compatibility with gas turbine engines. However, severe vibrations have been encountered which may cause spring failure or other damage to the assembly.

The vibrations may occur partially due to the difficulty in balancing the input and output clutch members. Generally, the input and output members are placed in an inboard/outboard relation, with the output member having the input member internally located, providing a series type spring mounting. While
45 both the input and output clutch members are dynamically and statically balanced prior to assembly, each will still have a slight residual imbalance. Once assembled, the input and output clutch members, coupled by the coil spring are dynamically balanced. However, each engagement and disengagement of the clutch spring with the driven member realigns the residual imbalances which could eventually result in the residual imbalances coinciding, providing an intolerable imbalance which generates severe vibration. Should the
50 imbalance occur in the drive mode, with the spring in the expanded condition, the spring coil which traverses the gap between the input and output members may partially enter the gap, misaligning or cocking the spring and causing strain with the potential for spring fracture.

The most serious problem encountered in high rpm operation occurs when the input member travels through a rigid body mode of vibration. A rigid body mode is so called because the assembly components
55 do not bend, but tend to vibrate as a spring/mass system about the supports. Balancing the components alone is insufficient to provide stable operation through a rigid body mode even if the point of maximum deflection is located at a bearing. Normally, rigid body modes can be traversed in operation by assuring that the parts are in good balance and by reliance on the damping inherent in a normal oil lubricated

bearing. However, it was found that the maximum deflection in a spring clutch assembly did not occur at the bearing locations, and could not be traversed without exceeding established G force limits, as the mode occurred at high speed and even with lubrication there was insufficient damping.

To stiffen the assembly to reduce deflection, the assembly may utilize increased spread between the bearings. However, this is difficult to accommodate where space is limited, and is also undesirable from a weight standpoint.

Another problem with existing spring clutches is attaining proper spring positioning. Generally, the spring is placed on the arbor using precision measurements, with the arbor then inserted and attached to the input member, usually by pinning. If the coil spring is improperly placed on the arbor, the spring may bind with the input member, inducing undesirable preload, which may fracture the end teaser coils during operation. Similarly, the absence of end spacing for accommodating axial spring displacement on the arbor could cause the spring to bind with the output member during expansion, causing improper clutch operation.

Yet another problem with clutch springs is the tendency to fracture in the transition area between the central coils and the end teaser coils. Generally, the difference in outside diameter of the teaser coils and central coils may cause one or more coils in the transition area to be unsupported when expanded, overstressing those coils and possibly inducing coil failure.

Typically, clutch springs have a central cross-over coil which traverses the gap between the input and output members formed by a near-axial cut during machining. This provides half the axial crossover coil in the input member and half in the output member. However, to achieve an axial cut using a conventional manufacturing method, a small hole is usually left on one side of the finished spring. Under the influence of centrifugal force, the missing material induces an imbalance, producing vibrations which could damage the spring clutch assembly.

When torque is applied to an overrunning clutch spring which does not have an axial cut, the spring may be unsupported as it pushes against the gap between the input and output clutch members. The spring then attempts to unwind into a U-shape and in so doing, may cause the input and output ends of the spring to be displaced radially with respect to each other, essentially cocking the spring. This radial deflection must be minimized to reduce or eliminate the resultant centrifugal imbalance which would be amplified in a high RPM application.

SUMMARY OF THE INVENTION

It is an object of the present invention to provide an overrunning spring clutch assembly which alleviates the difficulties in spring installation.

It is a further object of the present invention to provide a spring clutch assembly which is less susceptible to vibrations induced by residual imbalances.

It is a further object of the present invention to provide a spring clutch assembly which is usable at high RPM conditions, of about 30,000 RPM, with a reduced potential for spring failure.

It is another object of the present invention to provide a coil spring for a spring clutch assembly which eliminates the weakness between the end teaser coils and the center coils.

It is another object of the present invention to provide a coil spring which is less susceptible to imbalances induced by the cross-over coil cut.

According to the present invention, a spring clutch assembly for coupling a drive member to a driven member is disclosed which includes a housing, with an input clutch member and an output clutch member, each clutch member separably rotatably disposed in the housing. The assembly also includes an arbor extending interiorly through a bore in both members, the arbor engaged and rotatable with the input member. The assembly further includes a coil spring disposed over the arbor, about half the spring being in the input member and half being in the output member, the coil spring being expandable into contact with the input and output members for transmitting torque therebetween. The assembly further comprises an output member, placed in coaxial alignment with the input member, to provide a parallel mount and prevent residual imbalance superimposition, with a narrow gap separating the output member from the input member. The arbor further includes an extension which passes through the output member, and arbor bearing means, disposed in the housing, for rotatably supporting the arbor extension, thereby increasing the assembly bearing spread. The arbor may also include stop means, such as a shoulder, next to which the spring is placed, for assuring proper spring positioning within the assembly.

The extension of the arbor through the output end for support by the arbor bearing means increases the effective bearing spread of the input member/arbor subassembly, thereby stiffening the subassembly and reducing vibration. In addition, the overall configuration of the input and output clutch members is changed

from a series (inboard/outboard) to a parallel spring mount and residual imbalances are not superimposed. The stop means on the arbor facilitates proper axial alignment of the coil spring relative to the internal drive surfaces of the clutch members, thereby eliminating the concern over improper assembly.

The spring clutch assembly may also include an improved spring design which includes a cross-over coil which traverses the input to output member gap, the coil having a cut provided parallel to the spring axes, at a width of about 0,5 mm (0.020 inch), for restraining a greater portion of the medial coil section within the clutch members. The improved spring may also include a gradual ramp from the central coils to the teaser coils, extending over from 2-6 coils, to reduce the potential for a stress failure. The improved spring design may be provided, even if there are no arbor bearing means disposed in the housing for rotatable supporting the arbor.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a cross-sectional view of the spring clutch assembly of the present invention.

Fig. 2 is an enlarged view of a clutch spring usable in the assembly of Fig. 1.

Figs. 3A, 3B, 3C and 3D illustrate a first, rigid body mode shape and system natural frequencies for a spring clutch at various bearing spring rates.

Figs. 4A, 4B, 4C and 4D illustrate a second, first bending mode shape and natural frequencies for a spring clutch at various spring rates.

Fig. 5 is a graph of natural frequency versus spring rate for the input member bearings.

Figs. 6A, 6B, 6C and 6D are enlarged views of a prior art cross-over spring coil.

Figs. 7A and 7B are enlarged views of the inventive cross-over spring coil.

Fig. 8 is an enlarged view of a prior art central coil to teaser coil transition.

Fig. 9 is an enlarged view of a coil spring including a gradual ramp between the central and teaser coils.

Fig. 10 is another embodiment illustrating the gradual ramp between the central and teaser coils.

DETAILED DESCRIPTION OF THE INVENTION

Spring Clutch Assembly

Referring to Fig. 1, the spring clutch assembly of the present invention is shown. The assembly 1 includes a housing 2 which encloses an input clutch member 3 and an output clutch member 4. The housing, as shown, may be composed of various sections bolted together for ease in disassembly, but may also be of unitary construction. The input and output clutch members are generally cylindrical in shape and are disposed coaxially, and in end-to-end juxtaposition with a gap 5 therebetween. The input member is supported by bearings 6 disposed in the housing. The input clutch member includes a bore 7 axially disposed and open at an end thereof for positioning of a clutch spring 8 therein. The bore 7 provides an input drive surface 9 for spring engagement. A drive end 10 of the input member is engaged by a drive device (not shown) such as a gas turbine engine. The input member further includes a conical surface 11 interiorly thereof for mating with an arbor 12.

The output clutch member 4 includes a matching bore 13 for positioning of the clutch spring therein, with the bore 13 providing an output drive surface 14 for spring engagement. The output member 4 is rotatably supported by bearings 15 disposed within the housing 2. The output clutch member is engaged through a gear 16 to an output device (not shown) such as a gear box connected to a helicopter rotor.

The arbor 12 includes a shaft portion 17 with a conical end 18 for mating with the conical surface 11 in the input clutch member 3. The shaft 17 is hollow and includes a passage 19 which provides a flow path for lubricating oil. The conical surface 11 and conical end 18, are sized to provide a collet fit, maximizing radial support of the arbor. The collet fit increases the rigidity of the assembly by eliminating the slight gap required to fit pinned shafts together. Such rigidity is necessary to reduce centrifugal imbalances and limit vibrations.

The arbor shaft 17 extends from the input member 3 through the output clutch member 4, with the shaft 17 supporting the coil spring 8 placed thereover. The arbor further includes a shoulder 20, which is formed from the back wall of the conical end 18. The shoulder 20 provides a stop for precisely positioning the spring on the input side of the clutch assembly. Of course, the stop could be located on either end of the shaft, depending on the preferred method for assembly, and may be independent of the conical end 18.

The arbor shaft 17, unlike prior designs, extends beyond the end of the spring, passing through the output member bore 13 into the housing 2, where it is supported by bearings 21. The addition of bearings

to support the arbor provides additional stiffening of the entire assembly by effectively increasing the bearing spread of the arbor/input member subassembly, reducing the potential for vibrations. In addition, such an extension achieves the increased bearing spread while maintaining the same overall assembly dimensions. Consequently, the inventive assembly may be adapted for use in applications where restrictive space and weight requirements are encountered. The arbor extension also eases removal by providing means for gripping and withdrawing the arbor/spring subassembly from the input member.

The coil spring 8 is usually pressed onto the arbor, abutting the shoulder 20, and is thus fixed at the input end of the arbor to assure rotation therewith. Referring to Fig. 2, the coil spring 8 includes input teaser coils 22, output teaser coils 23, central coils 24, and a cross-over coil 25 which traverses the gap 5 between the input and output members. The teaser coils are of larger outside diameter than the central coils to assure continuous contact with the drive surfaces, 9 and 14, with the difference in diameter being about 0.5 mm (0.020 inch). The teaser coils 22 and 23 have oil relief grooves 26 to assure proper lubrication of the coils and drive surfaces.

In operation, a drive device, such as a gas turbine engine, turns the input clutch member 3, arbor 12 and coil spring 8 as a unit. Since the coil spring is fixedly engaged to the arbor, the spinning action causes the output teaser coils 23 to rub against the output drive surface 14, causing the spring to unwind and expand the central coils 24 outwardly into engagement with the input and output drive surfaces 9 and 14, respectively. Sufficient force is exerted to cause frictional engagement and consequently torque is transmitted from the input to the output clutch member. Should an autorotation be required, for example, when the engine rpm drops below the rotor rpm, the output member will run ahead of the turning coil spring, driving the teaser coils 23 to rewind the spring 8 and therefore disengage the output member 4 from the input member 3, preventing damage to the engine.

The clutch spring assembly of the present invention overcomes the problem of rigid body mode vibration. A critical speed analysis of the mode shapes and natural frequencies of a spring support system was performed using a Myklestad iteration technique. Myklestad's method is used for analyzing critical speed and deflection mode shapes in beams, with the beams represented by lumped masses connected by massless beam sections having bending stiffness. The procedure involves calculating a residual moment or deflection at the end of the beam which would sustain vibration at a selected frequency. Using an iteration process, the frequency that can be sustained with zero end moment can be determined, which is the critical or resonance speed.

Referring to Figs. 3A through 3D and 4A through 4D, the results of a critical speed analysis is shown for the driver/input member/arbor sub-assembly, represented at the top of the figures as a beam, with the input bearing represented as spring K. Mode shapes and frequencies are shown for various spring rates, with Figs. 3A through 3D showing the first, rigid body mode, and Figs. 4A through 4D showing the second, first bending mode of vibration. The results are presented in the following Table:

Table

FIG.	N/cm	Spring Rate, (lbs/in)	Rigid Body Mode, RPM	1st Bending Mode, RPM
3A/4A	8.850	(5,000)	5,500	30,500
3B/4B	88.500	(50,000)	13,700	31,000
3C/4C	265.500	(150,000)	20,200	32,000
3D/4D	885.000	(500,000)	25,200	33,600

The rigid body mode occurs between 5500 and 25,200 RPM, depending on spring rate, while the first bending mode occurs at or above around 30,000 RPM, regardless of spring rate. Therefore, the frequency of the first rigid body mode is highly influenced by the spring rate, whereas the frequency of the second mode (first bending mode) is more or less constant.

Fig. 5 shows the natural frequency versus the spring rate of the input member bearing, from 0 to 40,000 RPM. The rigid body mode is represented by line A, illustrating the relationship between mode occurrence and spring rate. The first bending mode is represented by line B, illustrating the relationship between mode occurrence and spring rate. A typical operating range for a helicopter is illustrated by bracket C, with idle at 10,000 RPM and a maximum speed of 30,000 RPM. From line A, it is seen that the rigid body mode may be moderated by adjusting the input bearing spring rate to be approximately 17,700 N/cm (10,000 lbs per inch) or less, preferably about 8.850 N/cm (5,000 lbs per inch), thereby traversing the mode during startup and shutdown only. Using a spring rate of 17,700 N/cm (10,000 lbs/in) or less completely eliminates the rigid body mode from the operating range. Of course, adjusting the spring rate to move the rigid body

mode above the operating range may also be used. For example, at the spring rate of about 885.000 N/cm (500,000 lbs/in), the rigid body mode occurs at about 25,000 RPM, and this could be extrapolated to place the rigid body mode above 30,000 RPM. However, it is preferable to reduce rather than increase the spring rate.

Adjustment of the spring rate is achieved in the present invention by using a thin film of oil between the input bearing outer race and the housing to effect a lower spring rate stiffness. The oil film produces a low spring rate damper which influences both the frequency and amplitude of vibration and preferably reduces the spring stiffness to less than about 10,000 lbs per inch. Accordingly, the low spring rate damper enables passage through the rigid body mode at low speed, below the projected operating range, while the inherent damping characteristics reduce the amplitudes of any encountered vibrations to enhance the service life of the clutch assembly.

Referring to Fig. 1, a squeeze film damper 27 is employed on an outer race 28 of the input bearings 6, forming a small clearance gap 29 between the race and the housing, for example, about 0,025 mm - 0,125 mm (0.001 - 0.005 inches), with grooves 30 for maintaining an oil layer in the gap. By varying the clearance or fluid viscosity, the effective spring rate and damping characteristics of the bearing is adjusted to the optimum setting. It may also be desirable to pressurize the oil to prevent fluid cavitation. Other spring rate damping means are usable with the present invention and may include an elastomer layer between the race and housing or a discontinuous contact race which allows the race to flex.

In high RPM applications, the spring clutch assembly has an increased sensitivity to axial alignment of the spring. Referring again to Fig. 1, a spacer 31 is provided on the arbor adjacent to the output member to provide a buffer of soft material between the end of the spring and the output member. Axial spring location is controlled by abutting the spring up to the arbor shoulder during assembly. The spacer then works in conjunction with the shoulder 20 to limit axial spring movement, with the spacer preventing the spring from wandering axially toward the output side of the clutch during its excursions from the arbor (compressed state) to the drive surfaces (expanded state). If there is insufficient axial clearance, the teaser coils 23 will tend to bind up during coil expansion and jam the spring at the output end. On the other hand, if there is excessive clearance, the spring can slowly move axially and have its central cross-over coil misplaced with respect to the gap between the input and output members, which could cause misalignment of the spring upon engagement. Generally an axial clearance of from 0,76 to 1,27 mm (0.030 to 0.060 inches) is sufficient to prevent these problems from developing.

Coil Spring for a Spring Clutch Assembly

An improved spring may also be included in the present assembly, yet is also equally adaptable to other spring clutch assemblies. Referring to Figs. 6A and 6B, a prior art cross-over coil 32 is shown including an axial cross-over cut 33. The clutch spring has a central cross-over coil formed by a conventional machining process. A cutter 34 cuts through a cylinder 35 at a particular pitch to form the coils. When the cross-over coil is reached, the cutter is directed axially for a predetermined amount and then redirected to cut at a given pitch. The axial cross-over cut straddles the gap between the members, assuring support without twisting. However, a small hole 36 is left on one side of the finished spring, shown in Fig. 6B in the compressed state, which could imbalance the spring.

Referring to Figs. 6C and 6D, a cross-over coil 37 is shown having a minimized cutout 38. A cutter 39 cuts through a cylinder 40 at a particular pitch to form the spring coils. When the cross-over coil is reached, the cutter is directed at a modified pitch chosen to minimize the cutout while supporting the spring in the gap. However, while the cutout is minimized, the angle formed would still tend to cock or misalign the spring during expansion, particularly in high RPM applications.

The clutch spring of the present invention overcomes this problem by including a cross-over coil having a very fine near axial cut, preferably produced using electron discharge machining (EDM) with a fine wire. Referring to Fig. 7, a cross-over coil 41 is partially cut using a cutter 42 which cuts through the cylinder at a predetermined pitch. When the cross-over point is reached, illustrated by point A, the cutter is removed and repositioned for cutting the remaining coils from point B to the end. A thin axial cut, preferably made by EDM, is made between point A and point B. The wire diameter is chosen for producing a cut which is approximately 0,5 mm (0.020 inches) wide. Three separate cuts are made to produce a smooth transition between points A and B: a first lead-in cut 43, covering an arc of about 90 degrees, an axial cut 44, and a lead-out cut 45, again covering an arc of about 90 degrees. As shown in Fig. 7B, when compressed, the central cross-over coil 41 has a substantially smaller cutout 46 for reducing any imbalance, while retaining full support of the cross-over coil across the gap. Under the twisting action of applied torque, the input and output clutch members will therefore remain aligned.

To further prevent the spring from unwinding into the gap, the gap between the input and output members should be reduced to a minimum, consistent with oil flow requirements. Generally a gap of about 0,76 mm (0.030 inches) is preferred.

Another problem with existing spring coils is that at the intersection of the teaser coils and the central coils. A sharp transition occurs where the helix of the spring cuts across the intersection of the teaser coils and the central coils, leaving a knife edge which produces a stress riser. Referring to Fig. 8, a prior art spring 47 has four teaser coils 48 and central coils 49, with a sharp transition coil 50 therebetween, with a radius R. When the spring is expanded, the coil 50, just outside of the last teaser coil, is left partially unsupported, with the stress potentially causing spring failure. For example, it was estimated that a coil, not in contact with the drive surface along only a 90-degree arc, developed an additional bending stress of 345 MPa (50,000 psi).

To eliminate this problem, a gradual ramp from the teaser coils to the central coils is utilized. Generally, a ramp which smoothly joins the teaser coils to the central coils over a spread of from 2 to 6 coils, with about 4 coils preferred, may be used with each coil increasing incrementally in diameter, or having each coil including a tapered outer surface to achieve the smooth ramp. Such a ramp distributes the stress over several coils, reducing the potential for failure.

Referring to Fig. 9, a coil spring 51 has three teaser coils 52, central coils 53, and four transition coils, 54, which provide a gradual ramp between the teaser and central coils. To minimize stress, an S-shaped surface is prepared with a first convex radius, R_1 , provided over the first two coils, and a second concave radius, R_2 , provided over the second two coils. As shown, the radius R is distributed over the four coils, rather than concentrated at a single coil.

Referring to Fig. 10, a coil spring 55 has three teaser coils 56 and six transition coils 57, which have tapered outer surfaces to provide a gradual ramp 58 between the central coils and the teaser coils. It should be noted that these views are highly exaggerated for illustrative purposes, as the difference in outside diameter is about 0,5 mm (0.020 inch). While a simple modification, this change in design adds significantly to the life of a coil spring by eliminating a stress riser.

The spring clutch assembly of the present invention, utilizing coaxial clutch members and an increased bearing spread is less susceptible to vibration from imbalances, providing a more durable assembly even when vibrations occur. Utilizing an arbor including a positioning shoulder reduces the chances for improper assembly, while the extension of the arbor eases disassembly. Incorporating damper means on the input bearing reduces the spring rate, avoiding a rigid mode vibration, while damping any imbalances which do occur. Consequently, the spring clutch assembly of the present invention is particularly suited for high performance applications where vibration must be kept to a minimum.

In addition, utilizing a clutch spring including a gradual ramp to the teaser coils reduces the potential for a stress failure, with the use of an axial, fine cut at the cross-over coil reducing the potential for twisting or cocking during clutch operation. Consequently, a spring clutch assembly incorporating such a spring is more durable and less susceptible to damage than conventional spring clutch assemblies.

Claims

1. A spring clutch assembly comprising a housing (2), an input clutch member (3) rotatably disposed in the housing (2), an output clutch member (4), rotatably disposed in the housing (2), the output member (4) placed in coaxial alignment with the input member (3) to form a parallel mount, each member (3,4) being separably rotatable, a narrow gap (5) separating the output member (4) from the input member (3), an arbor (12), extending interiorly through both the input and output members (3,4), the arbor (12) fixedly engaged to the input member (3), arbor bearing means (21) for rotatably supporting the arbor (12), a coil spring (8), about half the coil spring (8) disposed in a bore (7) of the input member (3) and half disposed in a bore (13) of the output member (4), the coil spring (8) being expandable into contact with the input and output members (3,4) for transmitting torque therebetween, **characterized in that** the arbor bearing means (21) are disposed in the housing (2) and the coil spring (8) is disposed over the arbor (12).
2. The clutch assembly of claim 1, **characterized by** input bearing means (6) mounted in the housing (2) for rotatably supporting the input member (3), and

damping means integral with the input bearing means (6) for adjusting the spring rate of the input bearing means (6) in an amount sufficient to prevent a rigid body mode of vibration from occurring in a desired operating range of the assembly.

- 5 3. The assembly of claim 2,
characterized in that
the damper means comprise a squeeze film damper (27) for producing a spring rate of about 17.700 N/cm (10,000 lbs. per inch) or less.
- 10 4. The assembly of any one of claims 1 to 4,
characterized in that
the arbor (12) has a conical end (18) for engaging a cone-shaped recess in the input member (3).
- 15 5. The assembly of any one of claims 1 to 4,
characterized in that
the arbor (12) has stop means (20) disposed thereon for positioning the coil spring (8) on the arbor (12).
- 20 6. The assembly of claim 5,
characterized in that
the stop means comprise a shoulder (20) formed integrally with the arbor (12).
- 25 7. The assembly of any one of claims 1 to 6,
characterized by
a spacer (31) disposed in the output member bore (13) for limiting axial travel of the spring (8) on the arbor (12).
- 30 8. The assembly of claim 7,
characterized by
an axial clearance between the spring (8) and the spacer (31) of about 0,76 to 1,27 mm (0.03 to 0.06 inch).
- 35 9. The assembly of any one of claims 1 to 8,
characterized in that
the narrow gap (5) is about 0,76 mm (0.03 inch) wide.
- 40 10. The clutch assembly of any one of claims 1 to 9,
characterized by
the input clutch member (3) having the axial bore (7) extending therethrough, the bore (7) including an input drive surface (9), the output clutch member (4) having the axial bore (13) extending therethrough, the bore (13) including an output drive surface (14) and the clutch spring (8) being expandable into contact with the input and the output drive surfaces (9,14) for transmitting torque from the input to the output clutch member (3,4).
- 45 11. The assembly of any one of claims 1 to 10,
characterized in that
the arbor (12) extends from the output member (4) into the housing (2) in an amount sufficient to increase the assembly bearing spread.
- 50 12. A method for providing a spring clutch assembly comprising
providing a housing (2) rotatably disposing an output member (4) in the housing (2), the output member (4) placed in coaxial alignment with an input member (3) to form a parallel mount, each member being separably rotatable, a narrow gap (5) separating the output member (4) from the input member (3), extending an arbor (12) interiorly through both the input and output members (3,4), the arbor (12) fixedly engaged to the input member (3), disposing arbor bearing means (21) for rotatable supporting the arbor (12), disposing a coil spring (8) about half the coil spring (8) disposed in a bore (7) of the input member (3) and half disposed in a bore (13) of the output member (4), the coil spring (8) being expandable into contact with the input and output members (3,4) for transmitting torque therebetween,
- 55

characterized in that

the bearing means (21) are disposed in the housing (2) and the coil spring (8) is disposed over the arbor (12).

5 **Patentansprüche**

1. Eine Federkupplungsanordnung mit:
 einem Gehäuse (2);
 einem Kupplungseingangsteil (3), das drehbar in dem Gehäuse (2) angeordnet ist;
 10 einem Kupplungsausgangsteil (4), das drehbar in dem Gehäuse (2) angeordnet ist und in coaxialer Ausrichtung mit dem Eingangsteil (3) plaziert ist, um eine Parallelanordnung zu bilden, wobei jedes Teil (3,4) gesondert drehbar ist und ein enger Spalt (5) das Ausgangsteil (4) von dem Eingangsteil (3) trennt;
 einer Welle (12), die sich innen durch das Eingangs- und durch das Ausgangsteil (3,4) erstreckt und die
 15 mit dem Eingangsteil (3) fest in Eingriff ist;
 einer Wellenlagerungseinrichtung (21) zum drehbaren Abstützen der Welle (12);
 einer Schraubenfeder (8), die etwa zur Hälfte in einer Bohrung (7) des Eingangsteils (3) und zur Hälfte in einer Bohrung (13) des Ausgangsteils (4) angeordnet ist, wobei die Schraubenfeder (8) in Kontakt mit dem Eingangs- und dem Ausgangsteil (3,4) zum Übertragen von Drehmoment dazwischen dehnbar ist,
 20 **dadurch gekennzeichnet**,
 daß die Wellenlagerungseinrichtung (21) in dem Gehäuse (2) angeordnet ist und daß die Schraubenfeder (8) auf der Welle (12) angeordnet ist.
2. Kupplungsanordnung nach Anspruch 1,
 25 **gekennzeichnet durch**
 eine Eingangslagerungseinrichtung (6), die zum drehbaren Abstützen des Eingangsteils (3) in dem Gehäuse (2) angeordnet ist, und eine Dämpfungseinrichtung, die mit der Eingangslagerungseinrichtung (6) eine Einheit bildet, zum Einstellen der Federkonstante der Eingangslagerungseinrichtung (6) auf einen Wert, der ausreicht, das Auftreten einer Starrkörper-Schwingung in einem gewünschten Betriebsbereich der Anordnung zu verhindern.
3. Anordnung nach Anspruch 2,
dadurch gekennzeichnet,
 35 daß die Dämpfungseinrichtung einen dämpfenden Quetschfilm (27) zum Erzeugen einer Federkonstante von etwa 17.700 N/cm (10,000 lbs. per inch) oder weniger aufweist.
4. Anordnung nach irgendeinem der Ansprüche 1 bis 4,
dadurch gekennzeichnet,
 40 daß die Welle (12) ein konisches Ende (18) zum Eingriff mit einer konusförmigen Aufnahme in dem Eingangsteil (3) besitzt.
5. Anordnung nach irgendeinem der Ansprüche 1 bis 4,
dadurch gekennzeichnet,
 45 daß die Welle (12) eine daran angeordnete Anschlageinrichtung (20) zum Positionieren der Schraubenfeder (8) auf der Welle besitzt.
6. Anordnung nach Anspruch 5,
dadurch gekennzeichnet,
 50 daß die Anschlageinrichtung eine integral mit der Welle (12) ausgebildete Schulter aufweist.
7. Anordnung nach irgendeinem der Ansprüche 1 bis 6,
gekennzeichnet durch
 ein Abstandsstück (31), das zum Begrenzen axialer Bewegung der Feder (8) auf der Welle (12) in der Bohrung (13) des Ausgangsteils angeordnet ist.
8. Anordnung nach Anspruch 7,
gekennzeichnet durch
 55 ein axiales Spiel von etwa 0,76 bis 1,27 mm (0.03 bis 0.06 inch) zwischen der Feder (8) und dem

Abstandsstück (31).

9. Anordnung nach irgendeinem der Ansprüche 1 bis 8,
dadurch gekennzeichnet,
daß der enge Spalt (5) etwa 0,76 mm (0.03 inch) breit ist.

10. Kupplungsanordnung nach irgendeinem der Ansprüche 1 bis 9,
dadurch gekennzeichnet,
daß sich die axiale Bohrung (7) des Kupplungseingangsteils (3) durch dieses erstreckt und eine
Eingangsantreibfläche (9) aufweist; daß sich die axiale Bohrung (13) des Kupplungsausgangsteils (4)
durch dieses erstreckt und eine Ausgangsantreibfläche (14) aufweist; und daß die Kupplungsfeder (8),
die zum Übertragen von Drehmoment von dem Kupplungseingangs- zu dem Kupplungsausgangsteil
(3,4) in Kontakt mit der Eingangs- und der Ausgangsantreibfläche (9,14) dehnbar ist.

11. Anordnung nach irgendeinem der Ansprüche 1 bis 10,
dadurch gekennzeichnet,
daß sich die Welle (12) in einem zur Vergrößerung des Lagerungsabstands der Anordnung ausreichenden
Ausmaß von dem Ausgangsteil (4) in das Gehäuse (2) erstreckt.

12. Verfahren zum Schaffen einer Federkupplungsanordnung, welches aufweist:
Bereitstellen eines Gehäuses (2);
drehbares Anordnen eines Ausgangsteils (4) in dem Gehäuse (2) in koaxialer Ausrichtung mit einem
Eingangsteil (3), um eine Parallelanordnung zu bilden, wobei jedes Teil gesondert drehbar ist und ein
enger Spalt (5) das Ausgangsteil (4) von dem Eingangsteil (3) trennt;
Erstrecken einer Welle (12) innen durch das Eingangs- und durch das Ausgangsteil (3,4), wobei die
Welle (12) fest mit dem Eingangsteil (3) in Eingriff ist;
Anordnen einer Wellenlagerungseinrichtung (21) zum drehbaren Abstützen der Welle (12); und
Anordnen einer Schraubenfeder (8), die etwa zur Hälfte in einer Bohrung (7) des Eingangsteils (3) und
zur Hälfte in einer Bohrung (13) des Ausgangsteils (4) angeordnet ist, wobei die Schraubenfeder (8) in
Kontakt mit dem Eingangs- und dem Ausgangsteil (3,4) zum Übertragen von Drehmoment dazwischen
dehnbar ist,
dadurch gekennzeichnet,
daß die Lagerungseinrichtung (21) in dem Gehäuse (2) angeordnet wird und daß die Schraubenfeder
(8) auf der Welle (12) angeordnet wird.

Revendications

1. Un ensemble d'embrayage à ressort comprenant un carter (2), un organe d'embrayage d'entrée (3)
monté à rotation dans le carter (2),
un organe d'embrayage de sortie (4) monté à rotation dans le carter (2), l'organe de sortie (4) étant
placé en alignement coaxial avec l'organe d'entrée (3) pour former un montage parallèle, chaque
organe (3, 4) étant susceptible de tourner séparément, un passage étroit (5) séparant l'organe de sortie
(4) de l'organe d'entrée (3),
un arbre (12) s'étendant à l'intérieur des deux organes d'entrée et de sortie (3, 4), l'arbre (12) étant mis
en contact fixe avec l'organe d'entrée (3),
des moyens de palier d'arbre (21) pour supporter à rotation l'arbre (12),
un ressort hélicoïdal (8), à peu près une moitié du ressort hélicoïdal (8) étant disposée dans un alésage
(7) de l'organe d'entrée (3) et l'autre moitié étant disposée dans un alésage (13) de l'organe de sortie
(4), le ressort hélicoïdal (8) étant susceptible de s'expandre pour venir en contact avec les organes
d'entrée et de sortie (3, 4), afin de transmettre un couple entre eux,
caractérisé en ce que
les moyens de palier d'arbre (21) sont disposés dans le carter (2) et le ressort hélicoïdal (8) est
disposé sur l'arbre (12).
2. L'ensemble d'embrayage sel on la revendication 1,
caractérisé par:
un moyen de palier d'entrée (6) monté dans le carter (2), pour supporter à rotation l'organe d'entrée (3)
et un moyen d'amortissement réalisé d'un seul tenant avec le moyen de palier d'entrée (6), pour

ajuster la constante d'élasticité du moyen de palier d'entrée (6), d'une valeur suffisante pour empêcher un mode de vibration de corps rigide de se produire dans une plage fonctionnelle désirée de l'ensemble.

- 5 3. L'ensemble selon la revendication 2
caractérisé en ce que:
le moyen d'amortissement comprend un amortisseur à film pincé (27) pour produire une constante d'élasticité d'à peu près 17.700 N/cm (10.000 lbs par pouce) ou moins.
- 10 4. L'ensemble sel on l'une quelconque des revendications 1 à 3,
caractérisé en ce que
l'arbre (12) présente une extrémité conique (18) pour venir en contact avec une cavité en forme de cône, ménagée dans l'organe d'entrée (3).
- 15 5. L'ensemble sel on l'une quelconque des revendications 1 à 4,
caractérisé en ce que
l'arbre (12) présente un moyen de butée (20) disposé sur lui pour positionner le ressort hélicoïdal (8) sur l'arbre (12).
- 20 6. L'ensemble selon la revendication 5,
caractérisé en ce que
le moyen de butée comprend un épaulement (20) réalisé d'un seul tenant avec l'arbre (12).
- 25 7. L'ensemble selon l'une quelconque des revendications 1 à 6,
caractérisé par
un élément d'écartement (31) disposé dans l'alésage (13) d'organe de sortie, pour limiter le déplacement axial du ressort (8) sur l'arbre (12).
- 30 8. L'ensemble selon la revendication 7,
caractérisé par
un passage axial compris entre le ressort (8) et l'élément d'écartement (31), d'à peu près 0,76 à 1,27 mm (0,03 à 0,06 pouce).
- 35 9. L'ensemble sel on l'une quelconque des revendications 1 à 8,
caractérisé en ce que
le passage étroit (5) présente une largeur d'à peu près 0,76 mm (0,03 pouce).
- 40 10. L'ensemble sel on l'une quelconque des revendications 1 à 9,
caractérisé par le fait que
l'organe d'embrayage d'entrée (3) présente l'alésage axial (7) le traversant, l'alésage (7) comprenant une surface d'entraînement d'entrée (9), l'organe d'embrayage de sortie (4) présente l'alésage axial (13) le traversant, l'alésage (13) comprenant une surface d'entraînement de sortie (14) et le ressort d'embrayage (8) est susceptible de s'expandre pour venir en contact avec les surfaces d'entraînement d'entrée et de sortie (9, 14), pour transmettre un couple de l'organe d'embrayage d'entrée (3) à celui de sortie (4).
- 45 11. L'ensemble sel on l'une quelconque des revendications 1 à 10,
caractérisé en ce que
l'arbre (12) s'étend depuis l'organe de sortie (4) pour aller dans le carter (2), d'une valeur suffisante pour augmenter l'écartement de palier d'ensemble.
- 50 12. Un procédé pour réaliser un ensemble d'embrayage à ressort, comprenant:
l'agencement d'un carter (2), le montage à rotation d'un organe de sortie (4) dans le carter (2), l'organe de sortie (4) étant placé en alignement coaxial par rapport à un organe d'entrée (3), pour former un montage parallèle, chaque organe étant susceptible de tourner séparément, un passage étroit (5) séparant l'organe de sortie (4) de l'organe d'entrée (3), l'insertion d'un arbre (22) à l'intérieur à la fois des organes d'entrée et de sortie (3, 4), l'arbre (12) étant en contact fixe avec l'organe d'entrée (3), la disposition de moyens de palier d'arbre (21) pour supporter l'arbre (12) à rotation, la disposition d'un
- 55

ressort hélicoïdal (8), à peu près une moitié du ressort hélicoïdal (8) étant disposée dans un alésage (7) de l'organe d'entrée et l'autre moitié étant disposée dans un alésage (13) de l'organe de sortie (4), le ressort hélicoïdal (8) étant susceptible de s'expandre pour venir au contact des organes d'entrée et de sortie (3, 4), pour transmettre un couple entre eux,

caractérisé en ce que

les moyens de palier (21) sont disposés dans le carter (2) et le ressort hélicoïdal (8) est disposé sur l'arbre (12).

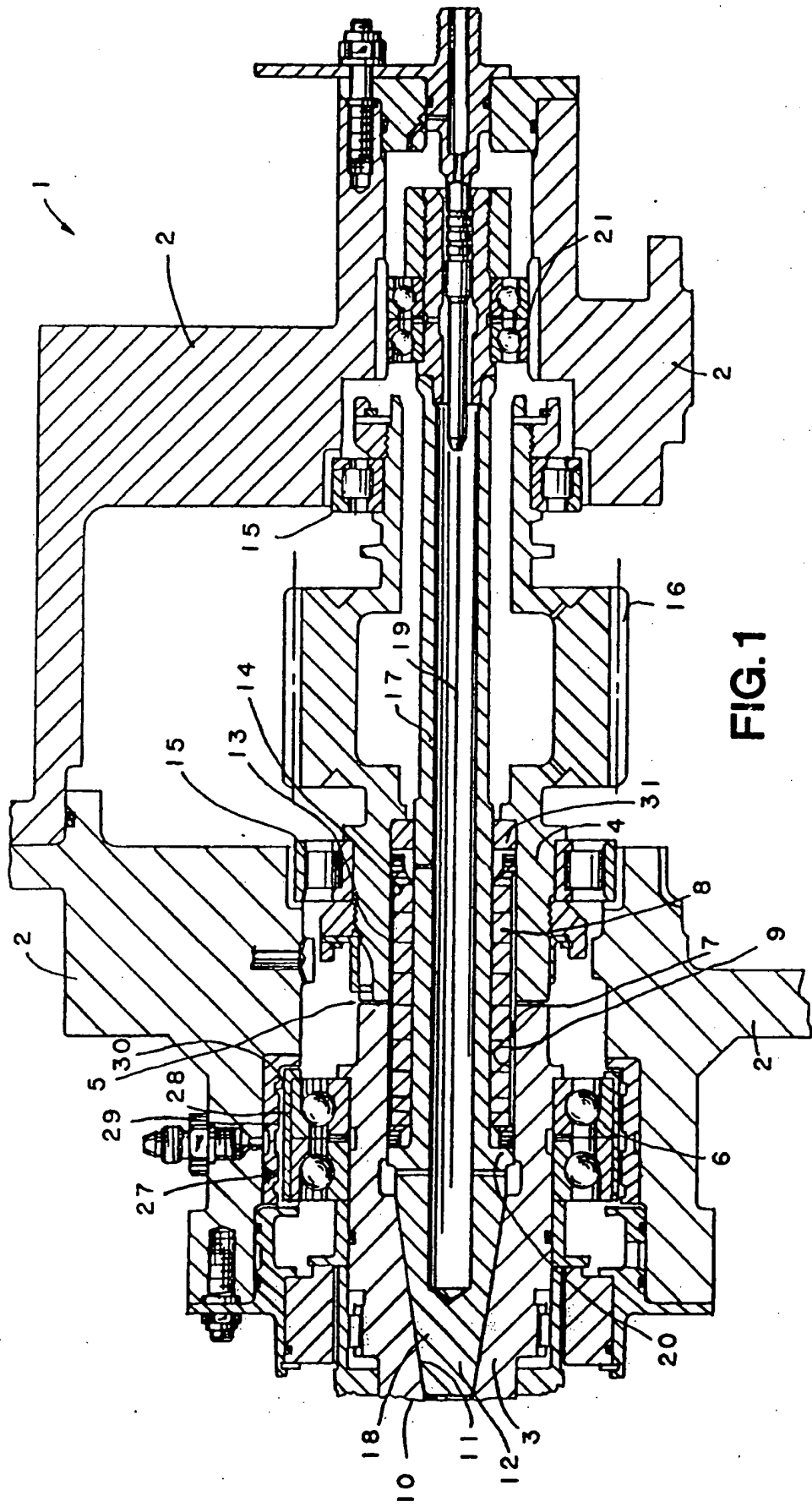


FIG.1

FIG.2

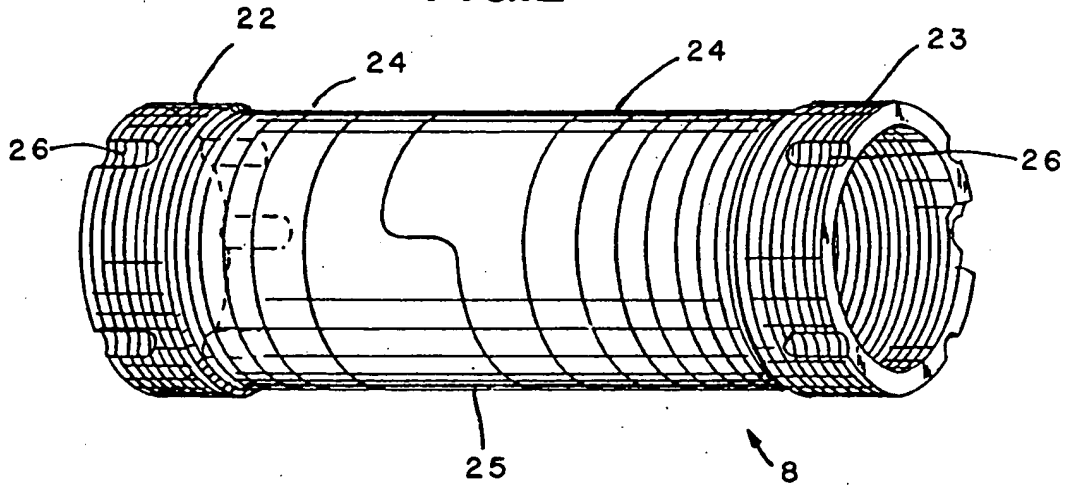


FIG.5

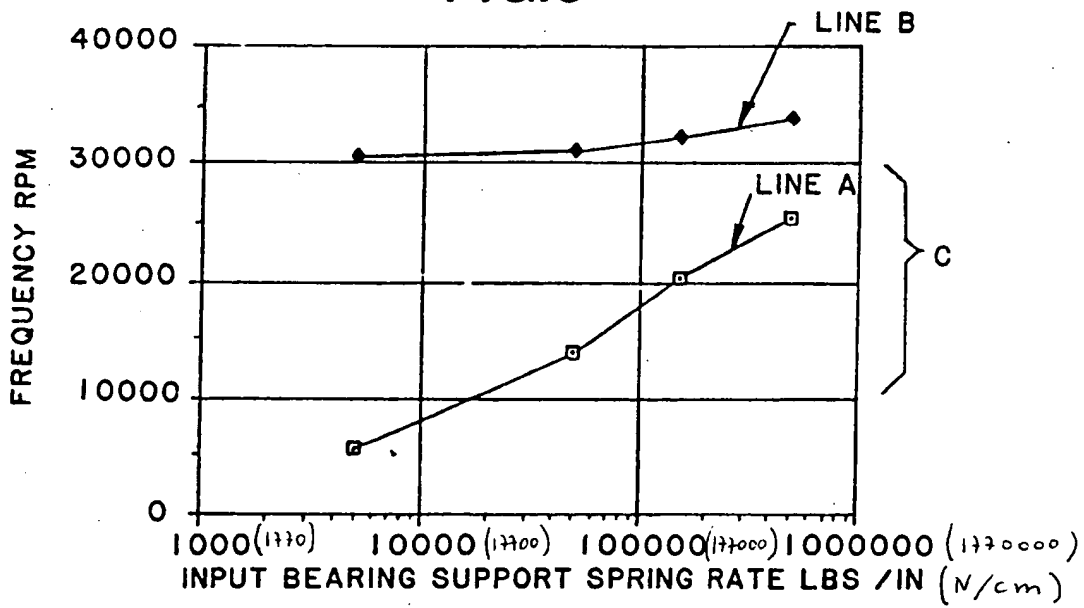


FIG.3A

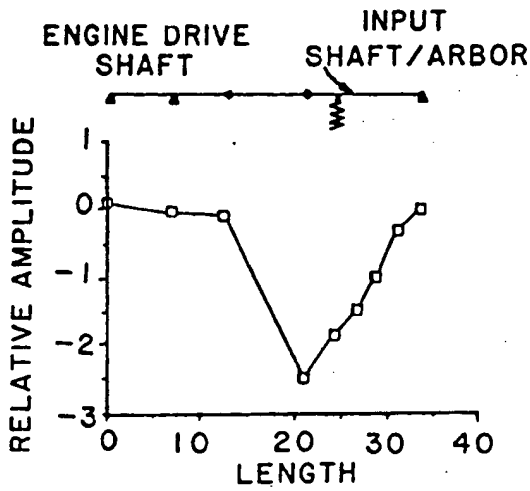


FIG.4A

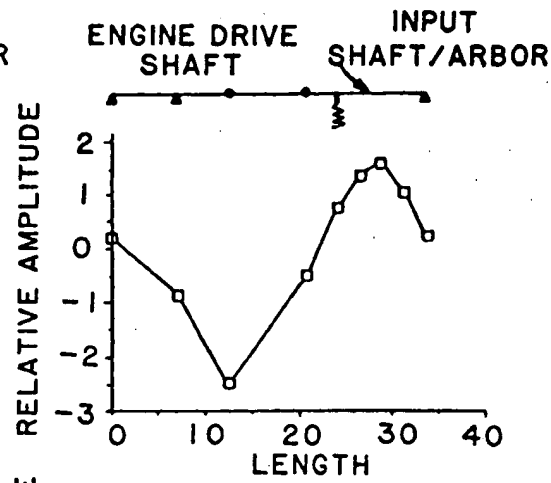


FIG.3B

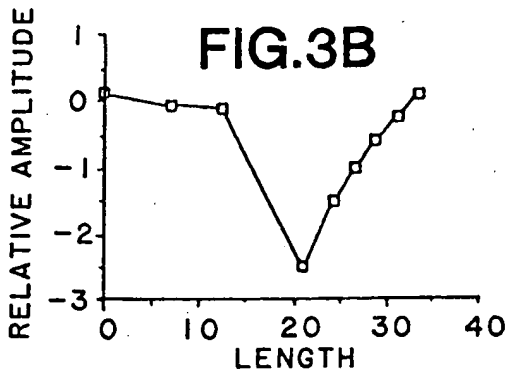


FIG.4B

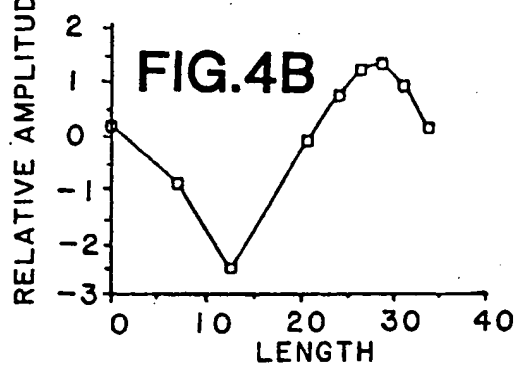


FIG.3C

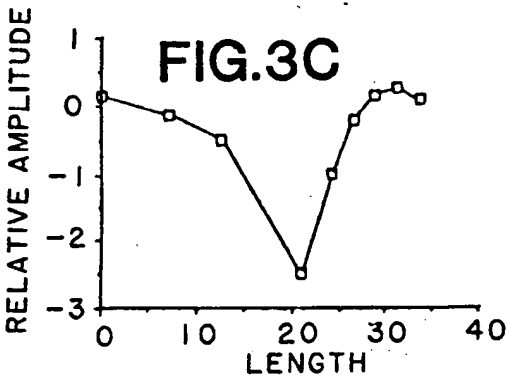


FIG.4C

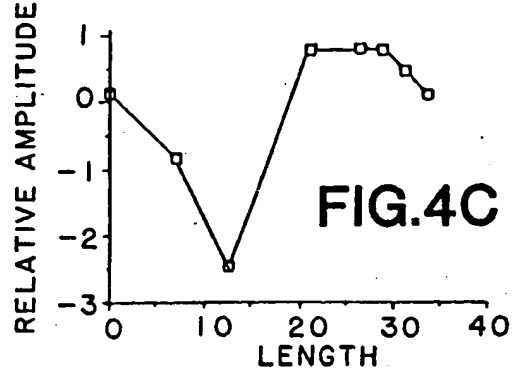


FIG.3D

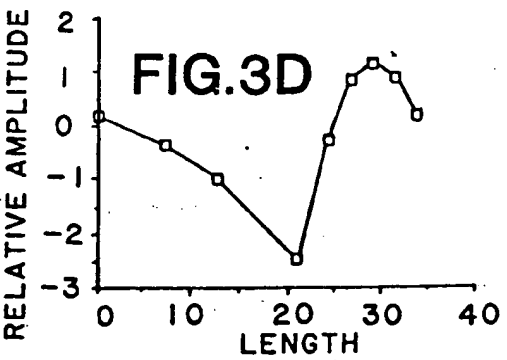
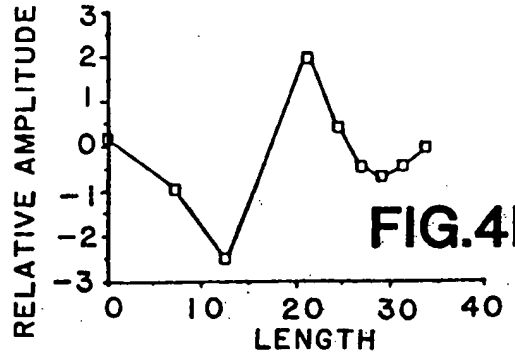


FIG.4D



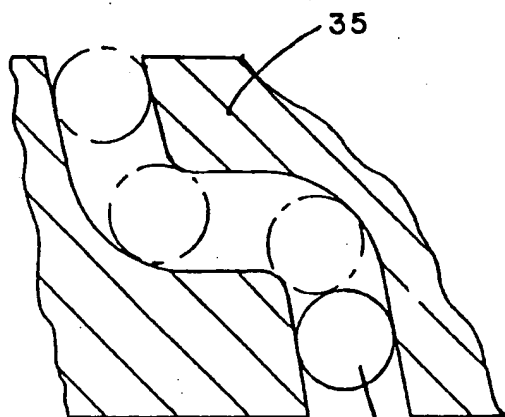


FIG. 6A
PRIOR ART

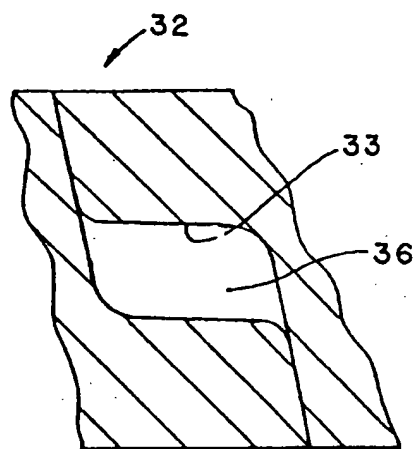


FIG. 6B
PRIOR ART

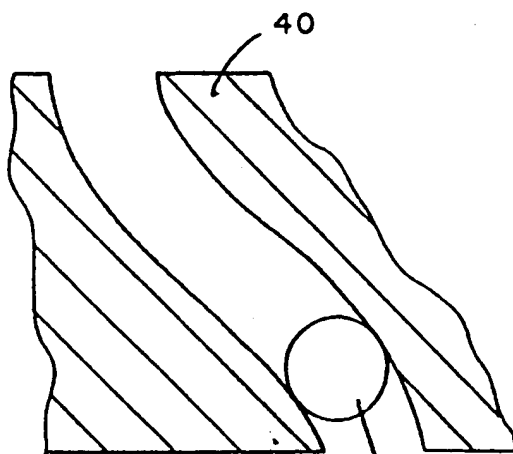


FIG. 6C
PRIOR ART

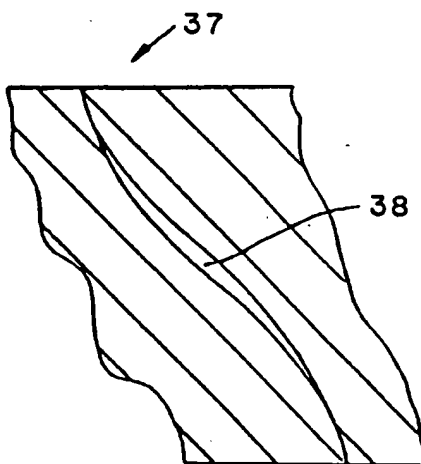
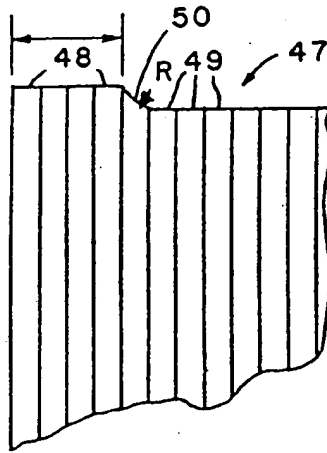
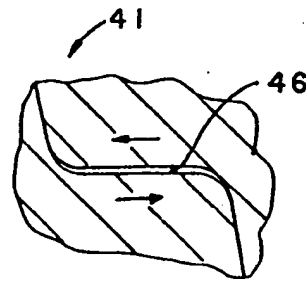
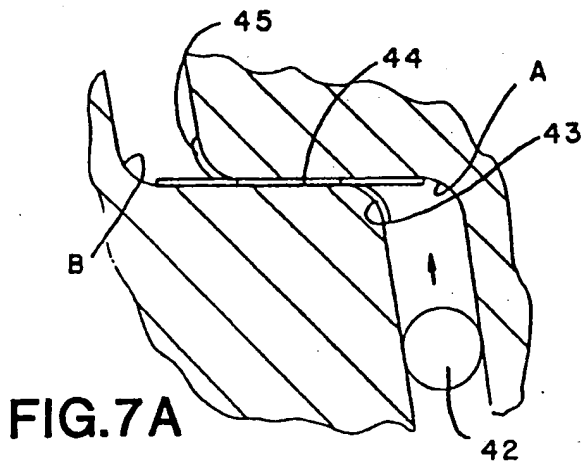


FIG. 6D
PRIOR ART



PRIOR ART

